

ENERGY RECOVERY IN A GAS DISTRIBUTION STATION BY COMBINATION OF EXPANDERS WITH PLANETARY GEAR AND OPTIMUM SINGLE VALVE CONTROL

by

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ABSTRACT

The prevailing use of expansion gases in distribution stations can be economically replaced by the application of turbo expanders. The advantage of turbine impulse wheels is that they have high efficiencies, even at very low partial load. With a single valve control easily programmable, seven valve points can be achieved with three control valves. Thereby, throttle losses can be minimized. The design of the valves and their drives can be adapted to the operation requirements. Bearings and couplings are deleted by a combination of expander and gear in one unit, resulting in less power loss, lower investments, and higher availability. The advantages of planetary gears compared to helicals are higher efficiency, smoother running, lower tooth loads, lower noise, and smaller plot plan. This results in secondary advantages such as smaller oil systems, baseplates, and foundations. The disadvantages are higher quality requirements regarding production and more components which, however, are smaller. When an automatic clutch is used, the LP section is shut down as soon as the gas mass flow is zero. The LP section is back into operation within a few seconds after having gas flow. The plant operates fully automatically without operators and maintenance personnel being present at the location.

The economic application of expansion turbines used for energy recovery depends not only on a turbine, but on all plant components. The gain of the useful energy is higher than the expenditure of investments, operating costs and maintenance costs.

The possibilities of energy recuperation from the natural gas pressure in distributing mains shall be demonstrated by means of an example.

The natural gas distribution station of a large German town is supplied from a 41 bar distribution main and transmits the natural gas in two networks of 13 bar or 5 bar. Previously, the natural gas was depressured through throttle valves, and the energy contained in the natural gas was destroyed. Now the gas expansion will be used for the generation of electric energy by the use of expansion turbines.

Technical plant data

Medium: Natural gas, molar mass	18.865	kg/kMol
Max. gas mass flow HP, LP	14.067	kg/s
Min. gas mass flow HP	approx. 2.8	kg/s
Min. gas mass flow LP	0	kg/s
Required control accuracy of the gas mass flow	+/- 0.5	%
Max. live gas pressure	48	bar
Live gas temperature - as delivered	5	C
Live gas temperature - expander inlet	75	C
Intermediate pressure piping	13	bar
Reheat temperature	75	C
Low pressure piping	5	bar
Expander speed	15.913	RPM
Max. generator output	4,100.	kVA
Generator speed	1,500.	RPM

Natural gas expansion is carried out in two stages. The HP stage reduces the pressure from 41 to 13 bar and the LP stage from 13 to 5 bar. After the HP stage an extraction range of between 0 and 100 percent is intended, meaning that the LP stage motive fluid ranges down to no flow. For this reason, both stages were designed as separate machines and arranged on each end of the generator (Figure 1).

During idling, the LP turbine impulse wheel would consume wheel friction and windage energy, therefore, it was connected to the generator through an automatic clutch. The clutch is torque controlled. It transmits the driving torque to the generator, but automatically disconnects as soon as the torque is negative.

For normal operation, the gas energy drop of the HP and LP section is approximately equal, as well as the maximum gas mass flow. Thus, both gears were selected to be identical, except for the opposite direction of rotation.

Energy flow

The maximum theoretical convertible working capacity of gas with isentropic expansion is called exergy [1]. For ideal gases it is:

$$hs = (k-1) / k * R * T1 * (1-(p1/p2)^{(k-1)/k}) \text{ in kJ/kg} \quad (1)$$

where

k is the isentr. exponent

R is the gas constant in kJ/kg/K

T1 is the inlet temperature in Kelvin

p1 is the inlet pressure in bar (abs)

p2 is the outlet pressure in bar (abs)

The power is

$$P = \dot{m} * h_s * \eta_s \quad \text{in kW} \quad (2)$$

where

\dot{m} is the massflow in kg/s

η_s is the isentropic efficiency

Heating the gas increases the exergy. Forty percent of the heating energy in HP stage is transferred to exergy. In the LP stage, the percentage is only 15 percent. Sixty percent of the heating energy of HP stage and 85 percent of the LP stage are nonconvertible energy.

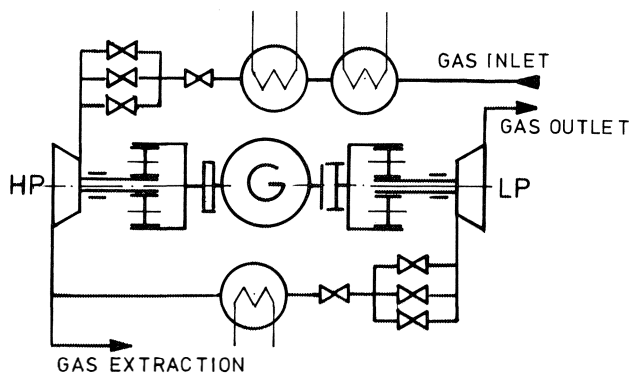


Figure 1. Arrangement of the Turbo Set.

The nonconvertible gas energy is called anergy. For the calculation of this plant, both terms are used as differences to the inlet condition (anergy = 0) and outlet condition (exergy = 0). The exergy of the gas with different plant positions is indicated (Table 1).

The values were represented in an exergy-anergy flow chart which is a modification of the Sankey chart (Figure 2).

The heating energy for heating the natural gas is waste heat. The heat of the first stage is the waste heat of the generator and gears. The heat transfer medium is water. The second heat exchanger and the reheater will be charged with heating water from the waste heat of another plant.

Table 1. Energy of natural gas during expansion.

Energy of natural gas during expansion	
Condition when delivered from distributing mains (41 bar, 5 C)	181.54 kJ/kg
Heat supply for heating to 75 C (energy)	160.09 kJ/kg
Inlet condition into the HP expander (41 bar, 75 C)	245.09 kJ/kg
Generated mechanical energy during expansion to 13 bar	133.11 kJ/kg
Condition when leaving the HP expander (41 bar, 12.5 C)	104.57 kJ/kg
Heat supply for heating to 75 C (energy)	133.47 kJ/kg
Inlet condition into the LP expander (12.5 bar, 75 C)	125.51 kJ/kg
Generated mechanical energy during expansion to 5 bar	100.34 kJ/kg
Condition when leaving the LP expander (5 bar, 24.3 C)	0.0 kJ/kg

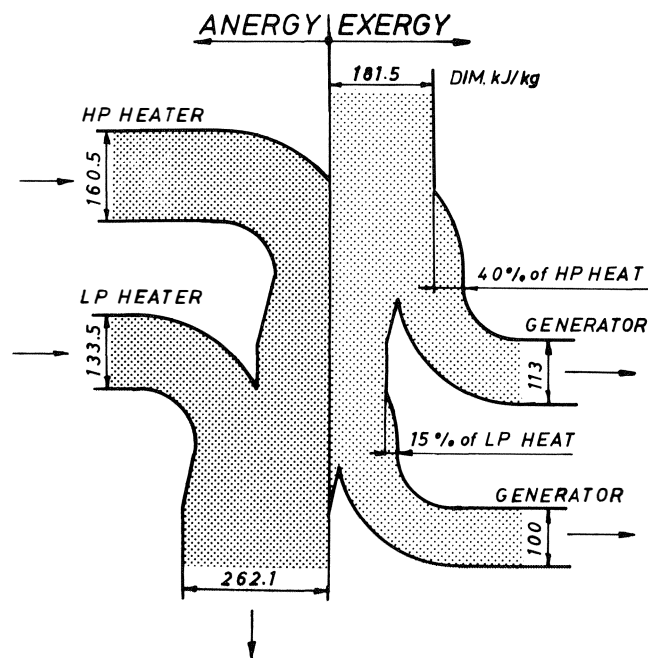


Figure 2. Exergy and Anergy Flow Chart.

Expander

As the best solution, the nearly constant drop of the natural gas with highly fluctuating gas flow results in a partially admitted impulse turbine wheel. Independent of the gas quantity, the blading can be dimensioned to an optimum in accordance with the pressure drop and the strength requirements. The blades of the impulse turbine wheels were machined from forged disks by means of electroerosion. The shroud bands were attached with high-strength ductile solder in vacuum to the blades. This results in a turbine impeller which, with regard to strength and vibration behavior, guarantees utmost safety and excellent availability. The nozzles are fabricated in the same manner. The impulse turbine wheel is axially flanged onto the gear pinion shaft by means of a Hirth gearing as commonly used in gear compressor engineering. This provides for easy dismantling of the turbine wheel for inspection of the shaft seal and the bearing. Owing to the precise assembly of the Hirth gearing, balancing after assembly can be deleted. The seal is an oil lubricated axial face seal which is successfully used in compressor engineering. The expander casings are welded barrel-type casings. They contain the intake nozzles with the nozzle groups, the outlet nozzles and the shaft seal to contain the natural gas from the atmosphere. The expander casings are flanged to the gear casings and do not have any bearings for the expander shaft. The bearings are contained in the gears. The components used are designed for several years of operation. Intermediate inspections are not required.

The bearing of the shaft are a tilting pad bearing and the planet wheel of the planetary gear, the latter being a latest innovation. The design has proven successful for compressor impellers, but it is new for turbines. The differences will be explained in the description of the planetary gear.

Mass Flow Control

The optimum efficiency of the impulse turbine wheel is only achieved when the gas flow can pass the nozzles nearly unthrot-

tled. This is only possible in the valve points which are characterized by fully open valves. Between the valve points, the control is carried out by throttling with control valves, whereby the losses are higher the larger the distance between the valve points. The number of nozzle groups cannot be increased as desired. In particular, the small expanders do not have enough space except for three or four nozzle groups. This problem can be profitably solved in connection with another problem.

Tested and approved control valves are required for expanders handling dangerous service gases. Standard turbine valves cannot be used and, thus, the standard valve control is not applicable. All valves may be controlled completely independent from one another at low expenditure when a programmable electronic control is used. However, when the valves are individually controlled, they can be opened and closed in any sequence. This provides for a minimization of the throttle losses. Seven different valve points can be achieved by means of three control valves, whereby one, two, or three control valves are fully open. The total operating range may be equally subdivided by a suitable selection of the nozzle group size. The total nozzle area is subdivided in the ratio 1:2:4.

One-seventh of the load can be operated unthrottled with the first nozzle group, 2/7 with the 2nd, 3/7 with both nozzle groups, and 4/7 with the 3rd nozzle group, etc.

The distance between two valve points is about 14.3 percent of the total load (Figure 3).

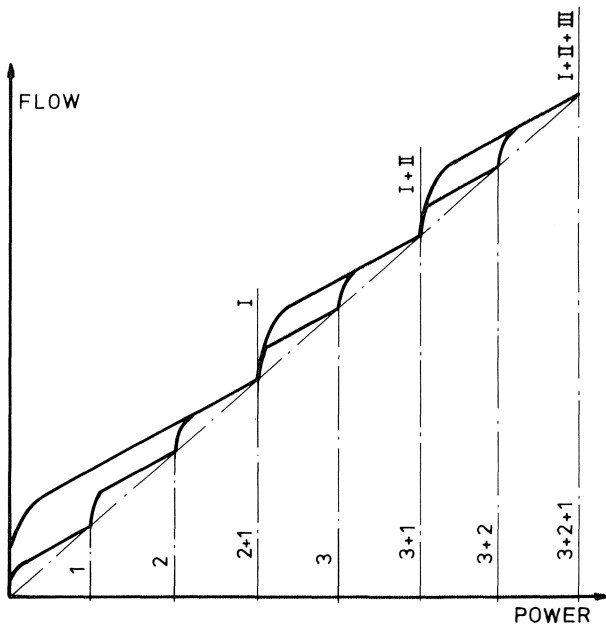


Figure 3. Effect of Valve Point on Power Output.

In general, smaller generators supplying a larger electric network can be operated without speed control after synchronization. Thereby, the control function is considerably simplified. Of course, there are different operating conditions when changing from one nozzle group to the next, and these conditions have to be thoroughly investigated. There are also different solutions, depending on the control function. In this case, the gas mass flow has to be controlled. The electric output of the generator along with the gas pressures and temperatures were not a control criteria.

The gas mass flow across the closing distance can be represented by the flow characteristics of the valves. If one assumes that the closing process takes place without control, it is neces-

sary to determine only the uncontrolled closing time of each valve. In this case, the valve to be opened is opened through an electronic ramp control in such a manner that the total of the gas mass flow remains constant (Figure 4).

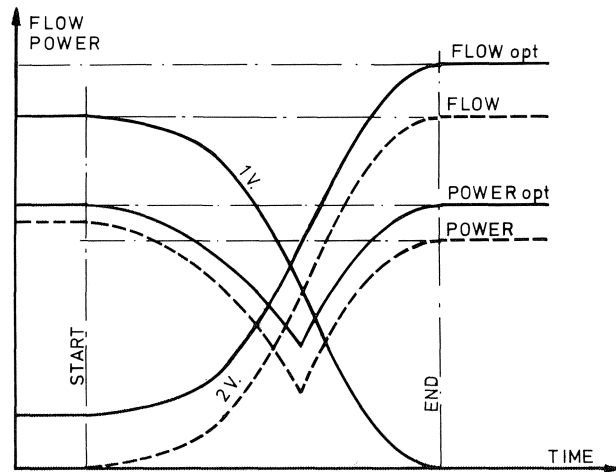


Figure 4. Changeover of First to Second Nozzle Group

The theoretical calculation does not cover all influences of the system, so that the control characteristic is to be subsequently optimized during operation. It is important that different ramp curves are prepared during load increase and decrease in order to eliminate valve oscillation in case the setpoint of the load coincides to a valve changeover.

In general, the changeover is caused by a setpoint variation, so that the changeover process takes place at higher setpoint/actual value deviations and does not fall into the tolerance range of 0.5 percent.

The process described above does not consider the power output. The optimum changeover points do not coincide with the starting points and final points of the control valves have to be determined.

This is explained by the example of changeover from the first to the second valve. When the first valve is fully controlled, an output is delivered which corresponds to the unthrottled gas flow. In case the first valve would be closed and the second valve would be opened with double the nozzle area on the same mass flow, the power output would be considerably lower. It would be more convenient to let the first valve open first of all and to let the higher gas mass flow past the second valve, even with lower power output but without output drop. The optimum time of changeover is only achieved when the total output from the first and second valve corresponds to the output of the second valve at equal mass flow. The optimum time can be determined from the performance characteristics.

In case the function of the output across the gas mass flow for two nozzle groups is known, the most effective output/gas mass flow ratio is known. This normally is the straight line from the capacity limit to the zero point. When this distance is entered into the output curve of the next group of stages, in such a manner that both end points of the distance coincide with the same increase in the curve, the upper final point of the distance indicates the optimum changeover point (Figure 5).

If the first valve has been fully opened, the second valve will be open until the total flow is

$$F_{2,2} = F_{1,1} + F_{2,1}$$

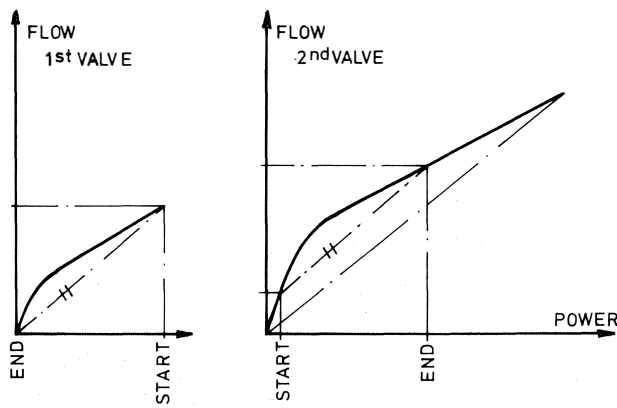


Figure 5. Optimal Changeover Point.

The power is:

$$P_{2,2} = P_1 + P_{2,1}$$

The flow power ratio decreases if the flow is larger than $F_{2,2}$, and the first valve is closed. The flow power ratio decreases also when the first valve is fully open and the second valve closes (dotted line in second valve diagram). The second valve should only operate in the range $R_{2,1}$ and $R_{2,2}$. The hatched area shows the saved flow at the same power or the higher power at the same flow.

During the changeover process, an unavoidable output drop also occurs here. The output after changeover, however, is the same as before changeover. The changeover process is the same as described above, although the opening of the next valve starts at the point which is marked by the lower final point of the distance.

The optimum changeover point for the third valve, which opens when the first and the second valve have already been controlled, is obtained in the same manner, except that the distance is obtained from the addition of the first two output curves.

With this plant, the control valves are pneumatically operated, turning control valves with a control accuracy of 1:100. The opening and closing times are less than five seconds. This is quite sufficient for the control function. In case of an electric load rejection, these setting times are insufficient in order to support the machine, because the flywheel effect of the expander is very low compared to the output. For this purpose, another pneumatically operated quickclosing valve is installed which closes immediately within 0.3 second by spring power in case the output drops. At full load of both machines, the expander speed increases by about five percent. After closing of the control valve and reduction of the speed to about 90 percent, the speed control is activated, and the turbo set is released for resynchronization. The synchronization takes place with the smallest valve with an output of maximum 14.3 percent of the total load, whereby the mechanical losses of generator and gear are to be achieved. Upon overcoming of the breakaway torque, 90 percent of the speed will be achieved within about 10 seconds. Thereafter, the generator will be excited and the speed slowly increased. The synchronization takes place automatically, provided number of revolutions, phase position and voltage are in compliance.

The LP expander will be run up in the same manner, except that the synchronization takes place through the clutch, when the synchronous speed is achieved.

Planetary Gears

In general, planetary gears are smaller than helicals of equal output. The forces of the teeth are distributed to at least three central meshings, so that they have smaller modules, smaller rolloff diameters and thus, lower circumferential speeds. The concentric action of the force considerably reduces the bearing loads, so that the bearings are designed smaller and operate at lower circumferential speeds. But, a larger number of bearing points and gearwheels are used. The disadvantage of the planetary gears are higher production precision and careful assembly. The advantages are higher efficiency with a smaller space requirement and lower quantity of oil required, as well as lower vibrations and noises. In many cases, it is advantageous that with planetary gears the shaft axis is not staggered. Planetary gears have proven successful for nonstationary plants (traffic areas). In stationary machine engineering, the advantages become more and more important. A considerable improvement is achieved with single-stage turbomachines by flanging the wheels to the pinion shaft of the planetary gear. The gear type coupling which previously was unavoidable is deleted and thus the oil supply of the coupling, tooth axial thrust, the possible vibration excitation, and production and assembly expenditure. The running bearing, including the necessary oil supply and the monitors are also deleted. The entire design will be more simple and well arranged (Figure 6). The question is whether the centering of the shaft in the planet wheel has the same damping and spring coefficient characteristics as a tilting pad bearing. As already known, the shaft of gear type couplings is centered by forces acting on the teeth as long as a torque is transmitted. The total tooth clearance is effective without torque.

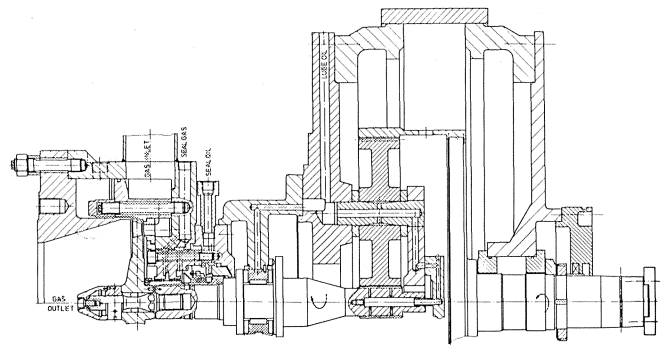


Figure 6. Planetary Gear with Integrated Expander.

The gear manufacturer has thoroughly checked the rotor dynamics by way of calculation with the result that (assuming a correct fabrication) the bearing in the planet wheel has a more stable running behavior than a sliding bearing. Although the calculation of the rotordynamics indicated satisfactory results, there were still remaining doubts whether all influences were correctly covered by the calculation. It was agreed to provide a guide bearing in addition to the tooth centering which would have to be installed if necessary. However, during trial run, it was determined that this preventive measure was not necessary. The centering effect of the teeth, in conjunction with the gyroscopic effect of the turbine impulse wheel and the damping through the planet wheel bearings, resulted in satisfactory running characteristics at all loads.

During the trial run, another advantage was noticed. During the trial run, the oil supply was inadvertently interrupted due to incorrect assembly and operation. The plain bearing, and particularly the thrust bearing and the shaft seal, showed the ex-

pected damage, while the teeth of the pinion shaft reacted similar to a rolling bearing and indicated no traces of damage at all. The gear itself was not damaged by this incident.

The bearing in the planet wheels was similar to that of a rolling bearing with high damping by the sliding grease film between the teeth and the planet wheels which are damped in their own bearings.

However because of the relatively high power density at the circumference, the pinion shaft reacted with vibration variations at load changes as can be noticed with large-scale machines of common design. Eccentric loads exist with partially admitted turbine impulse wheels. The output is concentrated on part of the wheel circumference and generates asymmetrical reaction forces in bearing and gearwheels, which actually should not occur in planetary gears. The dynamic behavior of the pinion shaft must change owing to the anisotropy of the sliding bearing and teeth. This was expected and acknowledged during operation.

Sometimes high axial forces occur with gear type couplings which are caused by a slight inclination in the teeth and a relatively high coefficient of friction (possibly even jamming). This phenomenon does not apply to the pinion bearing. Here the tooth profiles continuously slide on a fresh oil film. The maximum axial thrust which may be caused by the teeth corresponds to that of a normal gear.

The oil for the gear teeth is supplied from the center of the pinion shaft through radial bores. The oil sprayed into the central chamber of the pinion shaft is led to the teeth by means of centrifugal force effect.

The bearing of the planet wheel will be internally lubricated with oil through the planet wheel bolt. The clearance in the

planet wheel bearings can be kept relatively small. On one hand, the sliding speed is low, on the other hand, the planet wheel expands more during heating than the planet wheel bolt cooled by the oil and thus, causes a self regulation owing to the increase of the clearance and a higher oil throughput.

The structural design of the planetary gear is simple and sturdy. The lower bearing loads of the planetary gear compared to a helical can be particularly recognized on the narrow bearing of the low speed shaft. This optically demonstrates that the losses of a planetary gear are considerably lower than the losses of a helical gear, where the reaction forces of the toothings have to be completely carried by the bearings.

Because of the expected climatic changes caused by the increasing environmental load, the engineers are obliged to look for new methods and solutions for a better energy utilization. High technology and environmental protection are factors which complement each other and are not in contradiction.

The described plant is a prototype, with commissioning expected to be completed by mid 1990.

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